V. BARIČAK

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FLUID
FLOW WITHIN SLIDE BEARINGS - THE CAUSE OF SELF-EXCITING VIBRATION

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Due to rotor rotation fluid in bearing and/or seal is forced to rotate and rotating fluid force is generated. This force induce self-excited vibrations having very high amplitudes. Fluid induced self-excited vibrations model and their characteristics are discussed.

Key words: *self - exited vibration, dynamic stiffness, fluid whirl, fluid whip*

Tok fluida u kliznim ležajevima - uzročnik samouzbudnuh vibracija. Rotacijsko kretanje fluida u ležaju i/ili brtvenici pod djelovanjem rotacije rotora generira uzbudne sile koje dovode do razvoja samouzbudnih vibracija vrlo visokih amplituda. Prikazan je model nastanka ovih vibracija te njihove karakteristike.

Ključne riječi: *samopobudne vibracije, dinamička krutost, fluidni vrtlog, fluidni bič*

INTRODUCTION

This study research self-excited vibrations that are provoked by flow of fluid (fluid forces) in bearing and/or seal of rotating machines (all kind of machines that have sliding bearing: steam and gas turbine, rotation compressors, centrifugal pumps, generators, gear transfer).

This research includes description of the phenomenon and review of vibration characteristics.

SELF-EXCITED VIBRATIONS

Self-excited vibrations are developed when mechanical system includes internal or external source of energy, which gives energy equal or larger than energy loss [1-5]. The same source of energy, which keeps vibrations in order is permanent. The system makes sure that the energy inflow from permanent source through internal energy converter, which is a part of a feedback. Frequency of energy inflow responds to one of natural (its own) system frequency.

SELF-EXCITED
VIBRATIONS COUSED BY FLUID FLOW

Any dynamical system, including system of rotor-bearing/seal, includes certain stability sill (number of rotations) above which it becomes instable. Terms stability and sill of stability will be explained on example of isotropy rotor system, which include one radial mod and positive procession (Figure 1.). Mathematical model (differential equation) of movement of that kind of rotor look like following [6-8]:

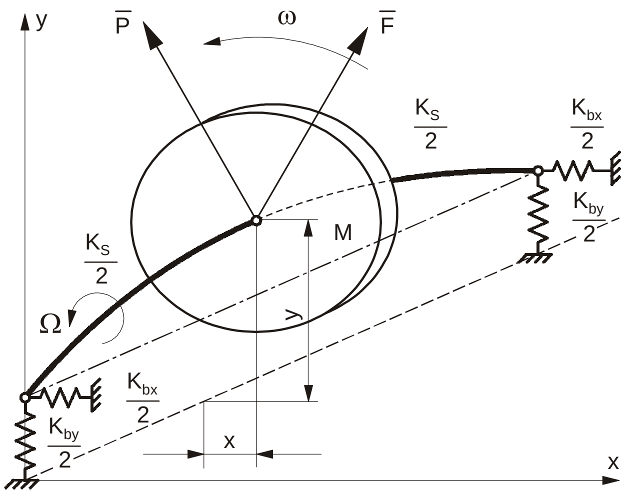

Figure 1. **Isotropy rotor system**
Slika 1. **Izotropni rotorski sustav**

$$M z + D_s z + K z + D (z - j \lambda \Omega z) = F e^{j(\omega t + \delta)} + P e^{j\gamma} \qquad (1)$$

Legend:

M = modal mass [kg]
D_s = damping of rotor surrounding [Ns/m]
K = modal stiffness [N/m]

V. Baričak, Faculty of Mechanical Engineering Tuzla, Tuzla, Bosnia and Herzegovina

D = fluid damping (e. g. in bearings, seal …) [Ns/m]
λ = proportion of average peripheral fluid speed (e. g. in bearings, seals …) [-]
Ω = peripheral of a rotor [s^{-1}]
δ = phase angle of excited force [°]
F = not synchronized rotation excited force [N]
P = static force [N]
z = $x + jy$ = radial coordination of rotor movement. It is consisted of movements toward axis X and Y

In physical shape, "*j*" presents motions toward axis *y* perpendicular on axis *x*. Also, physical "*j*" presents any action perpendicular on some other activity.

Model (1) is non-autonomy, linear differential equation that includes complex coordinate $z(t)$. This model describes forced vibrations. If excited force does not exist, that is if $F = 0$ and $P = 0$, model becomes autonomous and describe free vibrations:

$$M z + D_s z + K z + D (z - j \lambda \Omega z) = 0 \tag{2}$$

Results are $z = Ae^{st}$. Roots could be shown as $s_{1,2} = a_{1,2} \pm jb_{1,2}$.

Real parts of the root $a_{1,2}$ define fastness of decreasing (attenuation) of vibration amplitudes, and imaginative $jb_{1,2}$ its own system frequency. If $a_{1,2}$ are bigger than 0, amplitude grows exponentially. That means that stability condition is always $a_{1,2} < 0$.

Autonomous model predict sill of instability, which is deducted from the condition that second real part of the rood is equal zero (since one of them is always lower than zero). Sill of instability is rotation speed within witch free vibrations becomes harmonically with its own frequency $\sqrt{\frac{K}{M}}$ and constant amplitude *C*, and because of nullify of dumping force performed by tangential force $-j \lambda \omega D z$. It is defined by following:

$$\Omega = \frac{1}{\lambda} \frac{D_s + D}{D} \sqrt{\frac{K}{M}} \tag{3}$$

For rotation speed above Ω_{th} linear model predict exponential growth of amplitude of vibrations. It is not adequate any more, since in the practice it is not possible to reach endless growth of amplitudes. Nonlinear in the system limit the growth, so the model has to be supplemented by nonlinear terms.

Response of the rotor on rotation, periodical excited force is forced vibration frequency, which is adequate to frequency of excited force, and it is shown as:

$$z(t) = B\, e^{j(\Omega t + \beta)} \tag{4}$$

Response is late by the force for an angle, and could be shown as:

$$Be^{j\beta} = \frac{Fe^{j\delta}}{K - M\omega^2 + j[(D_s + D)\omega - D\lambda\Omega]} \tag{5}$$

Above-mentioned expression (5) is foundation of this analysis, since it defines that change of response could be challenged by the change of excited force or change of complex dynamics stiffness. However, expression in equation (5) presents complex dynamics stiffness that is characteristics of mechanical system to which the same is countered according to acting of excited force. Complex dynamics stiffness:

$$SDK = K - M\, \omega^2 + J\, [(D + D_s)\, \omega - \lambda\, \Omega\, D] \tag{6}$$

is consisted of direct term - direct dynamics stiffness (*IDK*):

$$IDK = K - M\, \omega^2 \tag{7}$$

and square dynamics stiffness *KDK*:

$$KDK = j\, [(D+D_s)\, \omega - \lambda\, \Omega\, D] \tag{8}$$

Complex dynamics stiffness is spring stiffness (*K*) of mechanical system, which include dynamics acting mass and damping. Complex dynamics stiffness is characteristic of system, which constrains to acting of dynamics forces (force) and make limit to vibration response of the system.

Strait (direct) dynamics stiffness is a component of complex dynamics stiffness of the system that collinearly opposing (rest on) to the acting of forces (force), and it is consistent of spring stiffness; modal mass and transversely connected damping term.

Square dynamics stiffness is a component complex dynamics stiffness of rotation system that is usually within fluid surrounding, and it is consisted of viscose damping $(D + D_s)\omega$ that present reaction of fluid when shaft is pressed on it and bearing part of fluid key $\lambda\Omega D$, which presents reaction of rotation shaft as a consequence of influence of fluid on it. Square dynamics stiffness generates reaction force perpendicular on force, which influence on rotor. Term $\lambda\Omega D$ comes from rotation characteristic of fluid dynamics force in bearing, and/or seal and present a key for stability of rotor system.

At the moment when sill of stability is overcome, when direct and square part of complex dynamics stiffness is equal zero, system became instable [7]. It resulted by self-excited vibrations. Within the work of rotation machines, the most often self-excited vibrations are caused by fluid forces, because of the fluid is consisted within two cylinders. One of those cylinders is steady, and the second is rotating by rotation speed Ω, and generally is processing by some process speed ω. Due to rotor speed, fluid is brought in peripheral velocity by some average peripheral speed λΩ (Figure 2.). The result is production of rotation fluid forces, and the tangential force is acting in sense of rotation and extremely destabilizing.

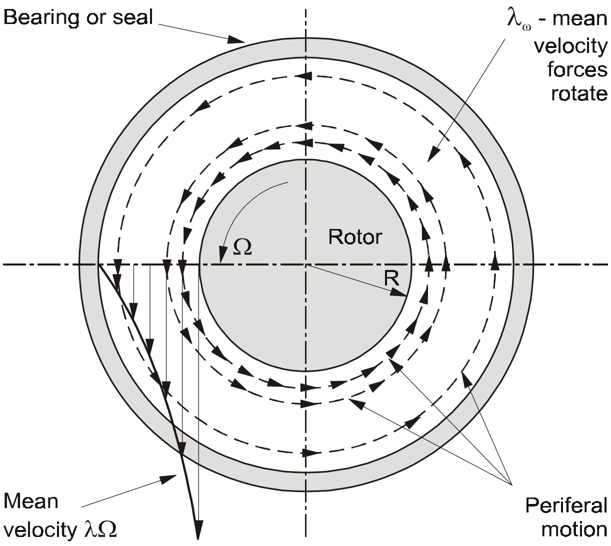

Figure 2. **Profile speed fluid**
Slika 2. **Profil brzine fluida**

The most often and the most famous self-excited vibration caused by fluids flow are famous as fluid whirl and fluid whip [8]. They are result of peripheral oil flow in slide bearing, media within compressor and pumps, steam within steam seal of steam turbines, and it is making very high amplitudes vibrations. Vibration characteristics of mentioned dynamics phenomena would be shown on following example, within which self-excited vibrations simulated dynamics model of the rotation machine of rotor kit made by Bently Nevada. Rotor kit is consisted of electromotor, one hydrodynamics and one lubricated bearing and rotor (shaft + disc). Number of rotations could vary from 0 up to 10 000 1/min. Constructed vibration probe are connected on interface DAIU-208P, which is sending data in computer. Software ADRE for Windows (Bently Nevada) provides differentiation of vibration signals in appropriate forms. Software supports 16 channels, and complete diagnosis equipment in on-line, which is providing follow up of dynamics behavior of machine within real time.

Rotor system within normal state and position (sleeves are in the state of resting and are placed in bottom part of the bearing) has 1. critical number of rotations on 1650 1/min. At the "filming" of vibration behavior of the model from 0 up to 8 000 1/min fluid forces caused by self-excited vibrations did not show. With support of spring frame the sleeve is placed in the central part of the bearing, and axes of bearing and sleeve were approximately the same. It means that off-centricity was equal zero. The model once again started from 0 up to 8 800 1/min. On 1 800 1/min harmonically vibration component is coming on whose frequency is ω = 0.32 X or ω = 0.32 Ω (Figure 3.).

This means that processing frequency is approximately equal to one third of rotation frequency. With growth of number of rotations, amplitude of vibration component, which is dominant in spectrum of vibrations, is growing. Relation between frequency of rotation and frequency of processing is constant up to 5 600 1/min, and its value is 0.32. With a further growth of number of rotations frequency of processing is not changed and its value is 27Hz, which is appropriate to "normal" resonant frequency of this model, if it include normal off-centricity (e = 0.4 + 0.6 usually). This resonant frequency is called it own frequency of high off-centricity, due to its own frequency on which starts self-excited vibrations caused by fluid forces (when e ≈ 0) that is called its own frequency of small off-centricity.

We could notice two vibrations of different characteristics of frequency:

1. the first, whose frequency follows up frequency of rotation whole time, and it is equal to mutilation of frequency of rotation and some number which is usually smaller than 1;
2. second, whose frequency is constant and is suitable to its own frequency.

To explain the difference between whirl and whip it is needed to remind on dynamics stiffness (terms 6, 7 and 8).

Spring stiffness is consisted of spring stiffness of a body (rotor) K_R and spring stiffness of fluid film of the bearing (bearings) K_B. This two stiffness are serial acting, so the total stiffness of K is reached from:

$$\frac{1}{K}=\frac{1}{K_R}+\frac{1}{K_B}\Rightarrow K=\frac{K_R\cdot K_B}{K_R+K_B} \tag{9}$$

From above written term it is clear that lower stiffness control total stiffness of the system, which is familiar from elementary theory of vibrations.

We will examine small off-centricity just like in above shown examination. During of creation of fluid whirl, rotor was working in the center of the bearing, so since off-centricity was small stiffness of oil film was small, or minimal. We are familiar that stiffness of the oil film is nonlinear function of off-centricity. Since spring stiffness of the oil film K_B was a lot lower since spring stiffness of the rotor K_R it control total spring stiffness of the system. Its own frequency was lowest possible and it was presented as $\omega_{\min}=\sqrt{\frac{K_B}{M}}$. At the same time value of its

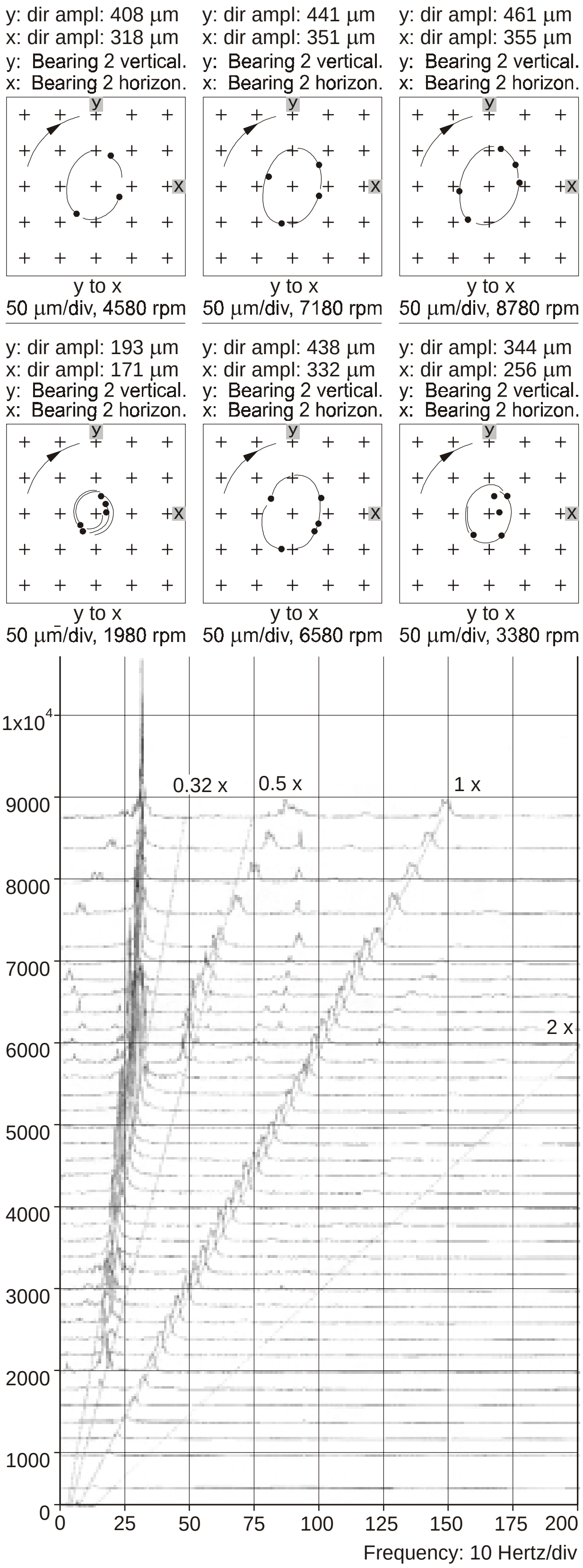

Figure 3. **Size eksperiments**
Slika 3. **Oblici eksperimenta**

own fluid frequency was $\omega_{fluid} = \lambda \times \Omega_{prag}$. At the creation of instability those two frequencies are the same, or $\omega_{min} = \omega_{fluid} \Rightarrow \sqrt{\frac{K_B}{M}} = \lambda \cdot \Omega_{prag}$. With creation of fluid stability on the sill of stability happens that proportion of the sleeve orbit (it is all about resonance, and amplitude is growing). Since orbit is growing, dynamics off-centricity is grooving too, and since spring stiffness of the oil film is nonlinear function of off-centricity, grows dynamics stiffness K_B oil film. Growth of K_B makes its own frequency higher, and at that moment system is going out of resonance. Increasing of stiffness K_B keeps rotor on the limit of stability and do not allow orbit increasing. In that case there exist different between zero direct and square dynamics stiffness. Within this case the frequency is 0.32X, or 0.32 Ω. 1X means frequency of vibration (processing), which is appropriate to rotation frequency. In that case processing is always positive, and in rotation direction.

Based on described it is possible to give definition of the oil whirl that is as following:

Oil whirl is positive circle proportion of the rotor including frequency, which is a part of a rotation frequency (usually 0.3 up to 0.49 Ω) and growing amplitude that is caused by growing number of rotations. At some number of rotations direct stiffness of the oil film K_B becomes so big that it is not any more the weakest stiffness in the system, and it becomes bigger than spring stiffness of the rotor ($K_B > K_R$). Spring stiffness of the rotor K_R is now the weakest in the system and it controls complete spring stiffness. Of course, it is not changed by a number of rotations, as stiffness of the oil film. Its own frequency is now equal to $\omega = \sqrt{\frac{K_R}{M}}$ and it is its own frequency of big off-centricity. Processing of rotor as a result of fluid instability asymptotically is closing through fluid whirl to the frequency, so processing frequency stays as a constant, without doubt on change of rotation number. This kind of processing is fluid whip. If a rotation number is higher so that system is going in some other mod, it is possible to do whole process again, which is that frequency of processing for some time became fraction of rotation speed and to cross in to frequency which is appropriate to other frequency of high off-centricity.

In the case of whirl mod processing of the rotor is the mod of stiff rotor, and in the case of whip mod it is similar to mod of first critical number of rotations (balance resonation). In the first case, sleeve within fluid bearing and central disk are at the phase, and in other case disk is late in phase after the sleeve for approximately 90 degrees. Shape of deflected axis of the rotor at the case of fluid whip clearly shows that at that time the same works were in resonance, and that amplitude of disk was likely larger than amplitude of the sleeve. This proportion sometimes could be 5:1. Since amplitude of the sleeve at the time of whip was approximately equal to air within bearing (approx. 0.2 up to 0.5 mm, depending on how big the machine is) it is clear that at that time amplitude of a central part of rotor are bigger than one millimeter, even up to several millimeters. It necessarily brings in to connection rotor and some parts of stator including all difficult consequences for the machine.

Theoretically, at the time of instability amplitude of vibrations (processing) would endlessly increase. Several times we mentioned that in the practice it is not happening because of no linearity within the system. Rotor is finishing within limited cycle of self-excited vibrations.

Balance of linear and nonlinear forces in the system defines the cycles.

CONCLUSION

Dynamics of rotor is a very complex area of general dynamics which is intensively researched due to its extremely large significant. Requests for as larger as possible forces including as smaller as possible dimensions, and requests for as larger as possible degree of useful acting resulted as machines that are rotating with very large number of rotations (>50000 o/min) including not so many air inside. All of it created new requests to the dynamics of rotor, since rotor system has to work so that it ensure stabile, reliable and safe work of the machine, including as larger as possible availability.

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